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(54) **VEHICULAR BELT-DRIVEN CONTINUOUSLY VARIABLE TRANSMISSION AND CONTROL METHOD THEREOF**

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USPC **474/28**; 474/18

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USPC 474/11, 18, 28, 13, 45–50; 701/51, 701/53, 54
See application file for complete search history.

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(57) **ABSTRACT**

In a vehicular belt-driven continuously variable transmission, belt squeezing force is inhibited from becoming excessive and a safety factor, with respect to belt slip, of belt squeezing force applied to a transmission belt (**48**) is reduced to a value that is less than or equal to 1.5 by reducing a pressure receiving area (S_{OUT}) of an output side hydraulic cylinder (**46***c*). As a result, a centrifugal hydraulic pressure canceller chamber on a secondary pulley side (**46**) can be eliminated thus simplifying the structure of the vehicular belt-driven continuously variable transmission, while belt squeezing force can be appropriately controlled.

8 Claims, 7 Drawing Sheets

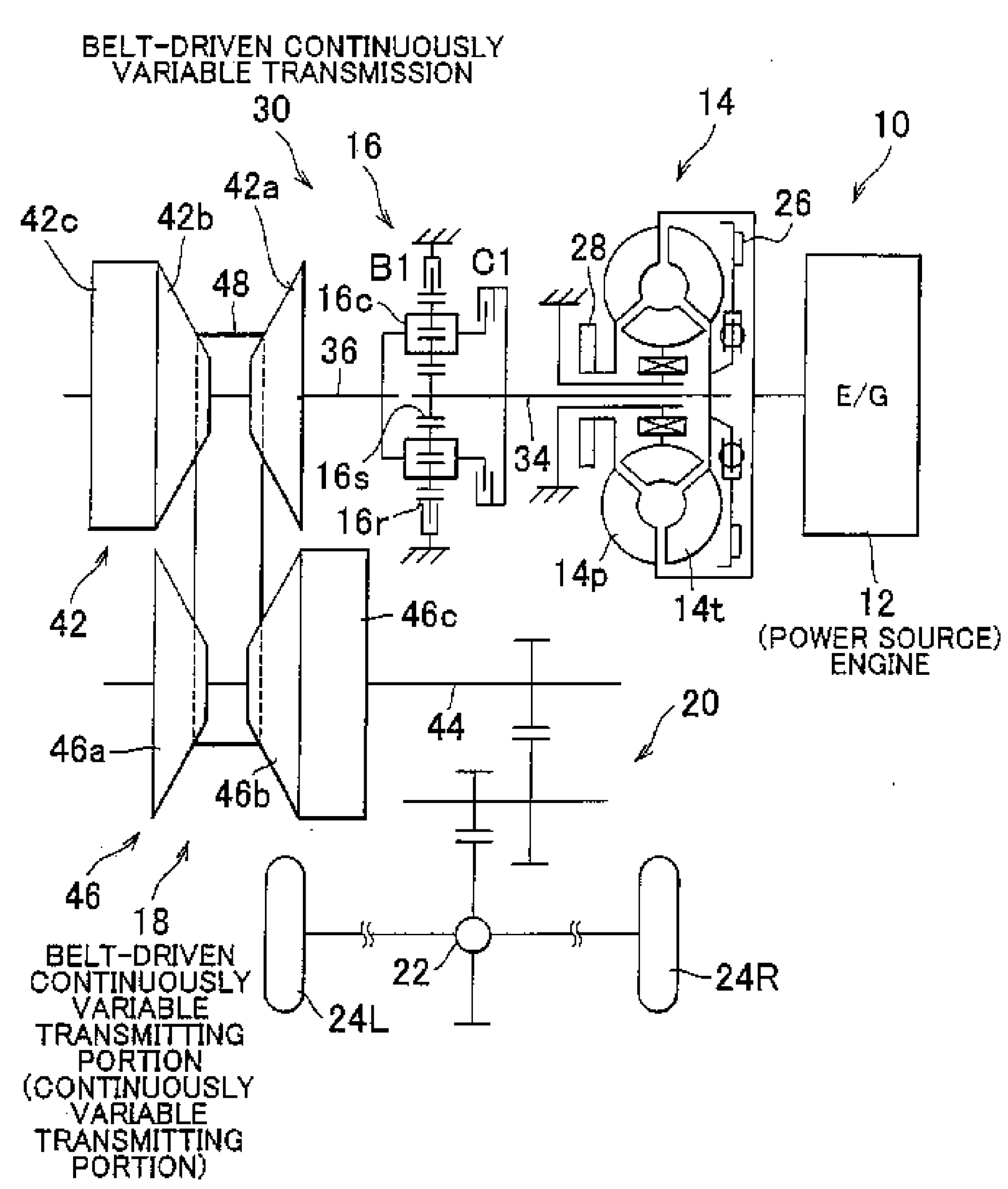

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FIG. 1

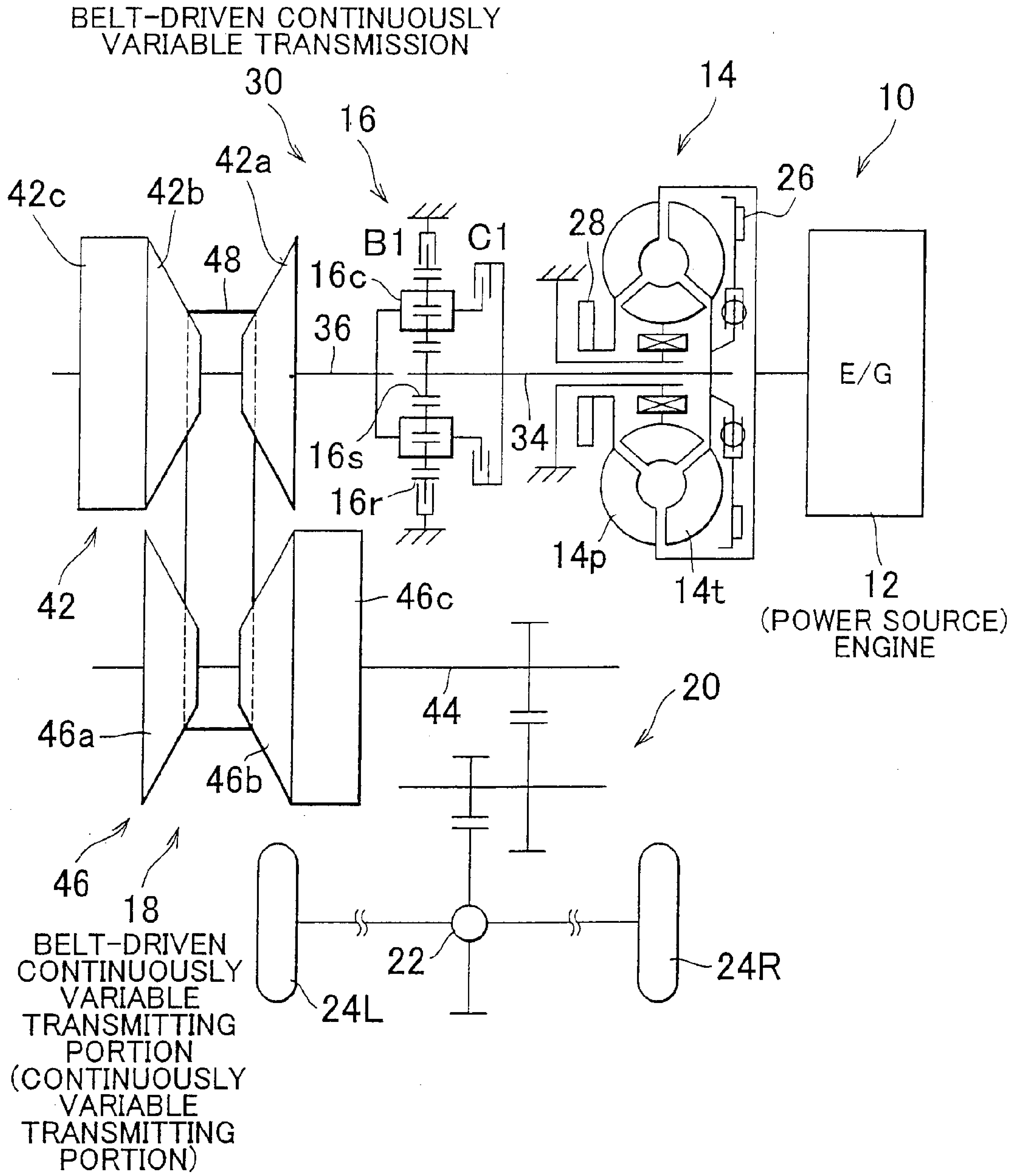

BELT-DRIVEN CONTINUOUSLY VARIABLE TRANSMISSION
30
16
14
10
42c
42b
42a
48
B1
C1
28
26
16c
36
E/G
16s
34
16r
14p
14t
12
(POWER SOURCE) ENGINE
42
46c
44
20
46a
46b
46
18
BELT-DRIVEN CONTINUOUSLY VARIABLE TRANSMITTING PORTION (CONTINUOUSLY VARIABLE TRANSMITTING PORTION)
22
24L
24R

FIG. 2

	C1 CLUTCH	B1 BRAKE
FORWARD	○	
REVERSE		○

○ : ENGAGED

FIG. 3

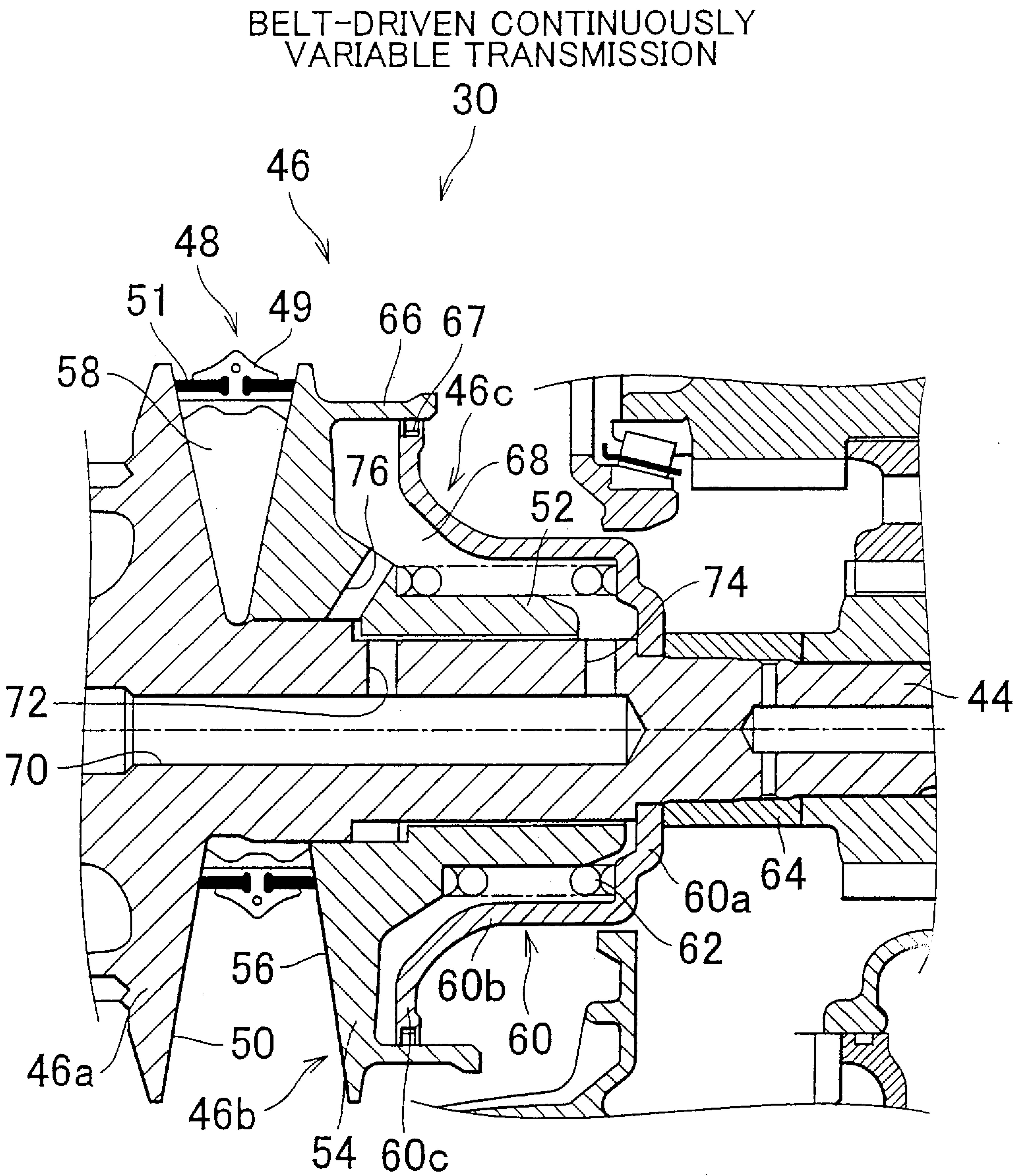

BELT-DRIVEN CONTINUOUSLY VARIABLE TRANSMISSION
30
46
48
51
49
66
67
58
46c
76
68
52
74
72
44
70
60a
64
62
56
60b
60
50
46a
46b
54
60c

FIG. 4

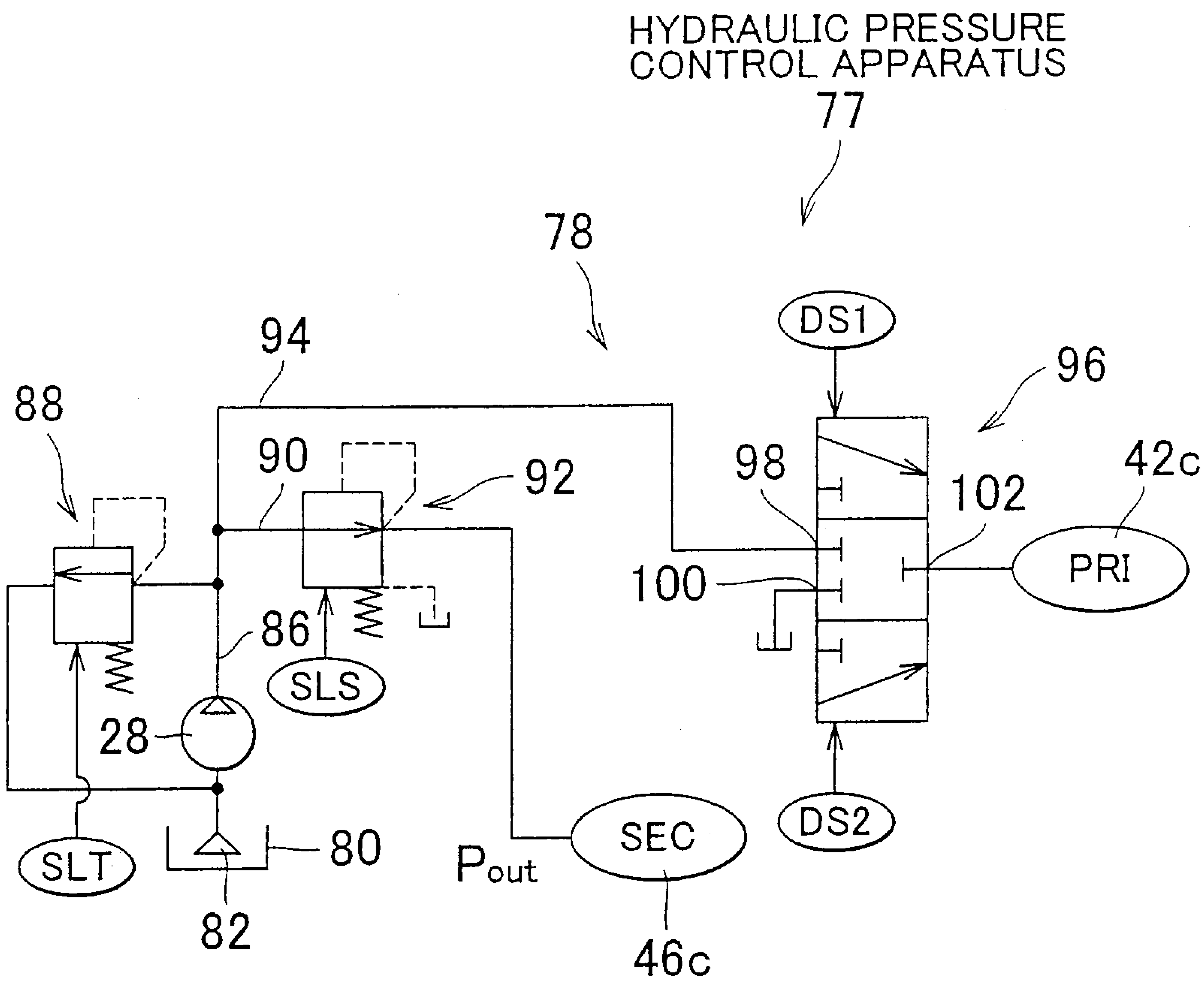

HYDRAULIC PRESSURE
CONTROL APPARATUS
77
78
94
88
90
92
DS1
96
98
102
42c
PRI
100
86
SLS
28
DS2
SEC
80
SLT
82
P_{out}
46c

FIG. 5

VEHICLE	HYDRAULIC PRESSURE CIRCUIT		CONTINUOUSLY VARIABLE TRANSMITTING PORTION SPECIFICATIONS						VEHICLE SPECIFICATIONS	
			Sec CYLINDER SPECIFICATIONS					BELT SPECIFI-CATIONS		
	INDICATED PRESSURE WHEN RUNNING AT MAXIMUM SPEED	MINIMUM CONTROLLABLE PRESSURE	SAFETY FACTOR AT MINIMUM PRESSURE	OUTER DIAMETER	INNER DIAMETER	PRESSURE RECEIVING AREA	CENTRIFUGAL HYDRAULIC PRESSURE COEFFICIENT WITH NO CANCELLER	WINDING DIAMETER	TORQUE	MAXIMUM VEHICLE SPEED
	[MPa]	[MPa]	[-]	[mm]	[mm]	[cm^2]	[$MPa/(km/h)^2$]	[mm]	[Nm]	[km/h]
VEHICLE OF THIS EXAMPLE EMBODIMENT	0.327	0.2	1.18	132.9	47.0	121.4	1.12E-08	129.3	161.5	209.6

FIG. 6

RELATED ART

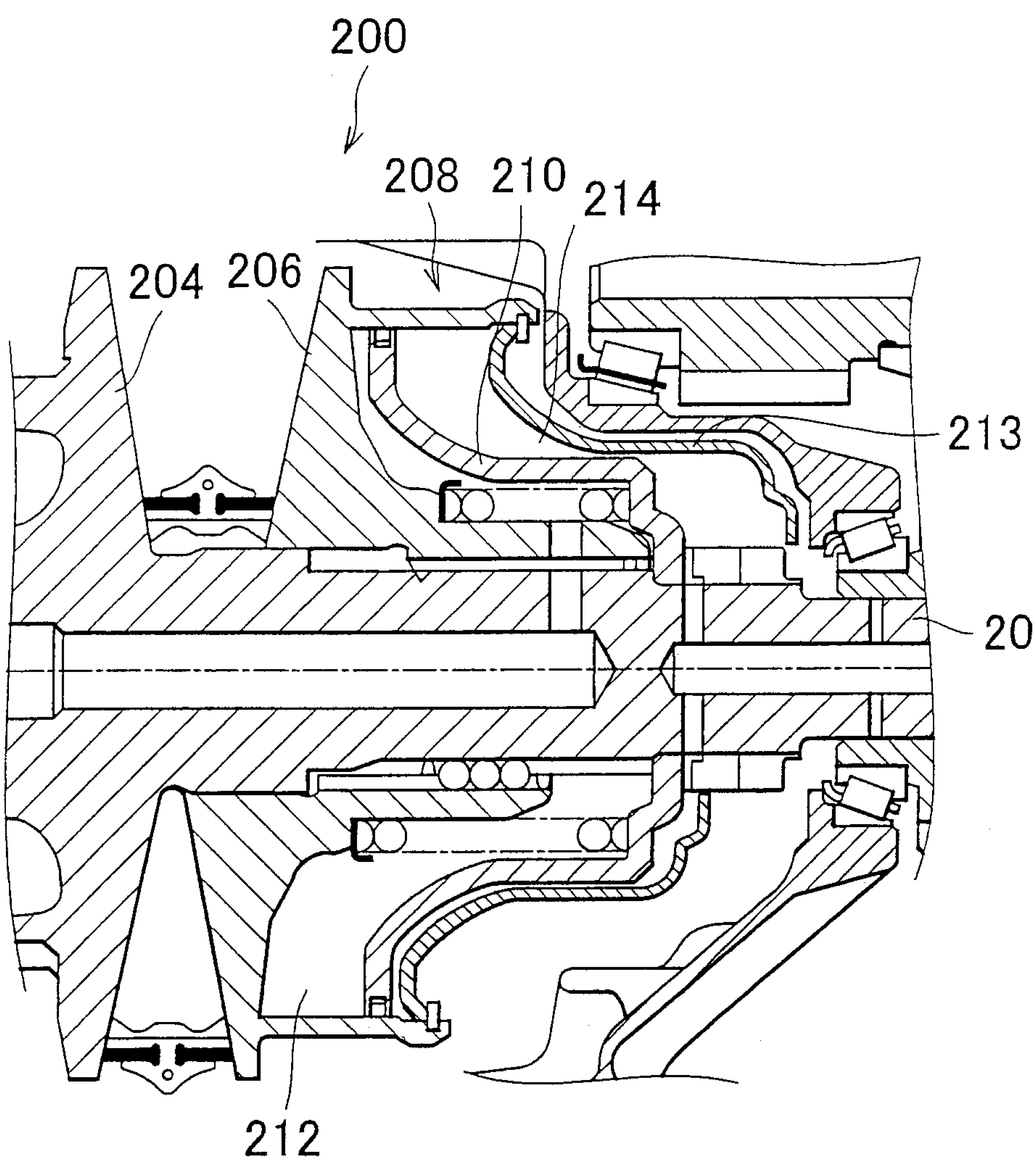
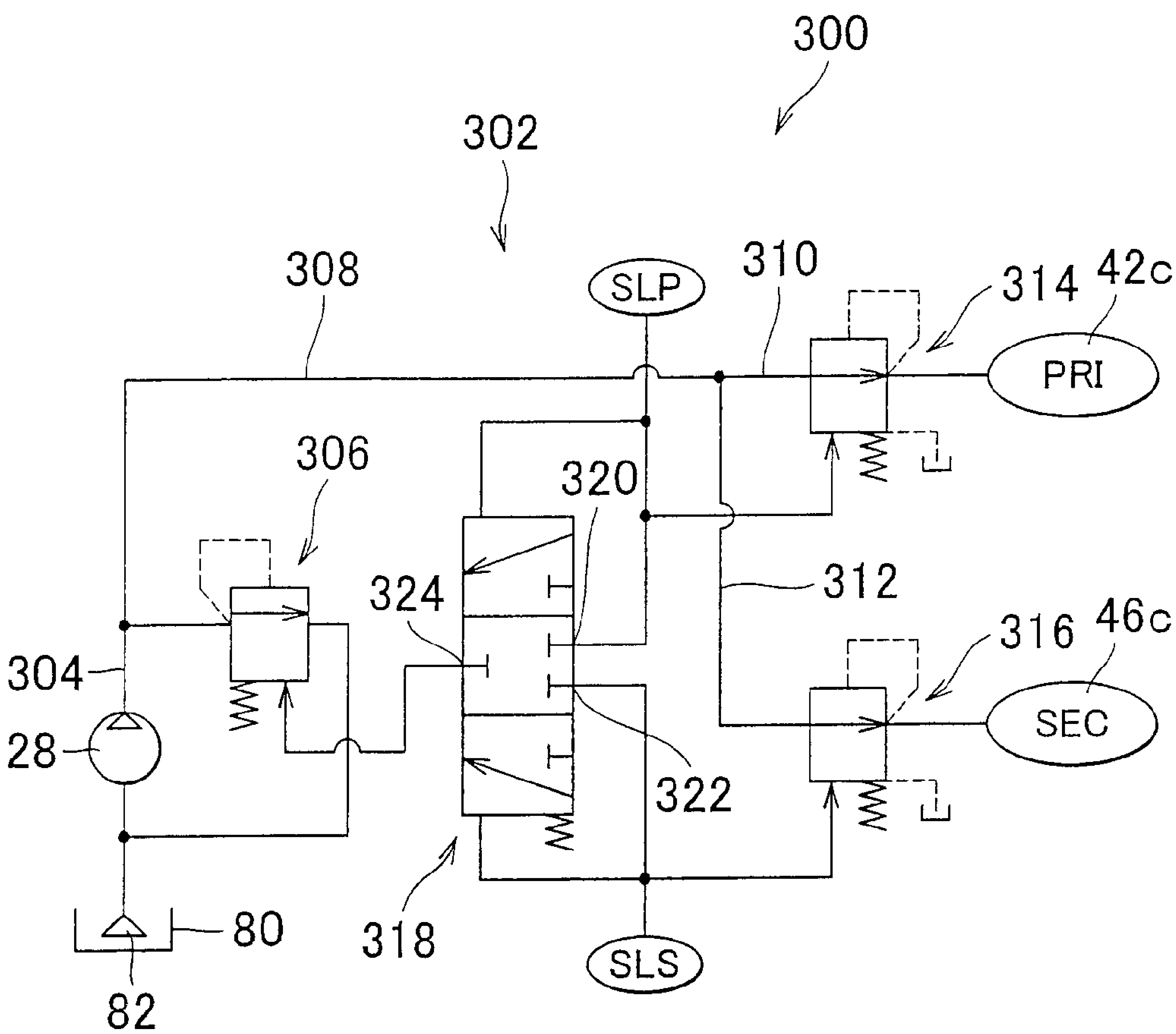


FIG. 7


300
302
308
SLP
310
314
42c
PRI
306
320
324
312
304
316
46c
SEC
28
322
80
318
SLS
82

VEHICULAR BELT-DRIVEN CONTINUOUSLY VARIABLE TRANSMISSION AND CONTROL METHOD THEREOF

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a national phase application of International Application No. PCT/IB2007/002777, filed Jun. 20, 2007, and claims the priority of Japanese Application No. 2006-188018, filed Jul. 7, 2006, the contents of both of which are incorporated herein by reference.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates to a structure and control method of a vehicular belt-driven continuously variable transmission. More particularly, the invention relates to the structure and control method of a vehicular belt-driven continuously variable transmission that eliminates a centrifugal hydraulic pressure canceller chamber of a secondary side cylinder.

2. Description of the Related Art

One known type of vehicular transmission is a belt-driven continuously variable transmission that shifts speeds smoothly and continuously without any gear switching. This belt-driven continuously variable transmission is formed of a continuously variable transmitting portion that mainly includes two rotating members arranged parallel to one another, a primary pulley provided on one of the rotating members so as not to be able to rotate relative to that rotating member, a secondary pulley provided on the other rotating member so as not to be able to rotate relative to that rotating member, and a belt that is wound around the two pulleys. The primary pulley and the secondary pulley each include a fixed sheave and a movable sheave, with a V-shaped groove in which the belt sits formed between the two. Power is transferred between the two pulleys via the belt. Here, a primary side cylinder, which applies thrust for moving the movable sheave of the primary pulley in the axial direction, is provided on the primary pulley, while a secondary side cylinder, which applies thrust for moving the movable sheave of the secondary pulley in the axial direction, is provided on the secondary pulley. By individually controlling the hydraulic pressure supplied to the primary side cylinder and the secondary side cylinder, the speed ratio of the belt-driven continuously variable transmission is changed by controlling the groove width of the primary pulley and changing the winding diameter of the belt around that pulley, while belt tension is controlled by changing the groove width of the secondary pulley.

In this kind of belt-driven continuously variable transmission, when rotation from a power source such as an engine is input to a continuously variable transmitting portion without a reduction in speed while the vehicle is traveling forward, the rotational speed of the secondary pulley increases, and as it does so, relatively large centrifugal hydraulic pressure is generated within the secondary side cylinder. This centrifugal hydraulic pressure applies thrust to the movable sheave of the secondary pulley in a direction that squeezes the belt such that the belt squeezing force becomes excessive. Because of this, one related belt-driven continuously variable transmission is provided with a centrifugal hydraulic pressure canceller chamber on the secondary pulley side to cancel out the centrifugal hydraulic pressure.

FIG. **6** is a sectional view of a secondary pulley **200**, which is a constituent member of the foregoing related belt-driven continuously variable transmission. The secondary pulley **200** includes a fixed sheave **204** integrally provided on an output shaft **202**, a movable sheave **206** fitted onto the output shaft **202** so as to be able to move in the axial direction but not rotate relative to that output shaft **202**, and a secondary side cylinder **208** provided adjacent to the movable sheave **206**. The secondary side cylinder **208** has a hydraulic pressure chamber **212** formed by the movable sheave **206** and a partition **210**, and a centrifugal hydraulic pressure canceller chamber **214** formed between the partition **210** and a peripheral wall **213** that is fixed to the movable sheave **206**. That is, the centrifugal hydraulic pressure canceller chamber **214** is formed on the opposite side of the partition **210** from the pressure chamber **212**. By providing the centrifugal hydraulic pressure canceller chamber **214**, a centrifugal hydraulic pressure equal to that in the hydraulic pressure chamber **212** is generated in the centrifugal hydraulic pressure canceller chamber **214** against the thrust on the movable sheave **206** toward the fixed sheave **204** which is generated by the centrifugal hydraulic pressure that is generated as the hydraulic pressure chamber **212** rotates. The centrifugal hydraulic pressure generated in this centrifugal hydraulic pressure canceller chamber **214** suppresses the effect of the centrifugal hydraulic pressure generated in the hydraulic pressure chamber **212** by applying thrust to the movable sheave **206** which is in the opposite direction as the thrust generated by the centrifugal hydraulic pressure in the hydraulic pressure chamber **212**.

Providing this centrifugal hydraulic pressure canceller chamber however makes the continuously variable transmitting portion heavier, less compact, and more expensive. Therefore, Japanese Patent Application Publication No. JP-A-2005-90719 describes technology which eliminates this centrifugal hydraulic pressure canceller chamber by forming the secondary side cylinder with two hydraulic pressure chambers, i.e., an outer diameter side hydraulic pressure chamber and an inner diameter side hydraulic pressure chamber, and appropriately switching the cylinder pressure receiving area.

However, with the technology described in JP-A-2005-90719, a structure is necessary to switch between a mode that supplies hydraulic pressure to the inner diameter side hydraulic pressure chamber and discharges hydraulic pressure from the outer diameter side hydraulic pressure chamber, and a mode that supplies hydraulic pressure to the inner diameter side hydraulic pressure chamber and also supplies hydraulic pressure to the outer diameter side hydraulic pressure chamber. However, the required structure is rather complex. Also, the belt squeezing force does not change smoothly as the cylinder pressure receiving area, which is related to the switching of the modes, is switched.

SUMMARY OF THE INVENTION

This invention thus provides a vehicular belt-driven continuously variable transmission that is able to appropriately control belt squeezing force while having a simplified structure realized by eliminating a centrifugal hydraulic pressure canceller chamber on a secondary pulley side.

A first aspect of the invention relates to a vehicular belt-driven continuously variable transmission that includes a) a continuously variable transmitting portion into which rotation from a power source is input without a reduction in speed while a vehicle is traveling forward, and a single hydraulic pressure chamber provided for a secondary pulley, and b) a hydraulic pressure control apparatus that shifts the continuously variable transmitting portion by controlling one of i) the supply and discharge of the hydraulic fluid and ii) the pressure of the hydraulic fluid with respect to a primary side cylinder provided for a primary pulley, and adjusts belt squeezing force of a belt wound around the primary pulley and the secondary pulley by controlling the pressure supplied to a secondary side cylinder formed of the single hydraulic pressure chamber. c) The hydraulic pressure control apparatus is structured to control a line pressure and the pressure supplied to the secondary side cylinder independently.

Also, in the first aspect, a cylinder pressure receiving area of the secondary side cylinder may be set such that a safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to a predetermined lowest controllable pressure when the vehicle is traveling at maximum speed on a flat road, is a value less than or equal to 1.5.

Accordingly, although the belt squeezing force would become excessive at maximum speed due to increased thrust generated by centrifugal hydraulic pressure pushing the movable sheave in the direction that increases the belt squeezing force because no centrifugal hydraulic pressure canceller chamber is provided to cancel out that thrust, that belt squeezing force can be inhibited from becoming excessive by reducing the pressure receiving area of the secondary side cylinder. In this case, it is necessary to also reduce the pressure supplied to the secondary side cylinder. Regarding this point, controlling the line pressure and the pressure supplied to the secondary side cylinder independently makes it possible to avoid problems such as the line pressure, together with the pressure supplied to the secondary side cylinder, becoming too low, or not being able to shift to increase the speed due to the hydraulic pressure necessary for the shift not being supplied to the primary side cylinder or the like. Also, when the pressure receiving area of the secondary side cylinder is reduced, the line pressure must be increased to increase the hydraulic pressure supplied to the secondary side cylinder at low running speeds. However, if the line pressure can be controlled independently from the pressure supplied to the secondary side cylinder, an increase in the line pressure can be limited to roughly the speed reduction side ($\gamma>1$) so adverse affects on practical fuel consumption can be avoided. Also, the cylinder pressure receiving area of the secondary side cylinder is reduced until the safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the secondary side cylinder is set to the lowest pressure during maximum speed flat road running, in which the vehicle is running at maximum speed on a flat road, becomes a value less than or equal to 1.5, which makes it possible to keep the durability of the belt from declining. As a result, it is possible to provide a vehicular belt-driven continuously variable transmission having a simple structure and enabling the centrifugal hydraulic pressure canceller chamber to be eliminated for all practical purposes.

A second aspect of the invention relates to a control method of a vehicular belt-driven continuously variable transmission that includes a continuously variable transmitting portion into which rotation from a power source is input without a reduction in speed while a vehicle is traveling forward, and a single hydraulic pressure chamber provided for a secondary pulley. This control method is characterized by including a) shifting the continuously variable transmitting portion by controlling one of i) the supply and discharge of a hydraulic fluid and ii) the pressure of the hydraulic fluid with respect to a primary side cylinder provided for a primary pulley; and adjusting belt squeezing force of a belt, which is wound around the primary pulley and the secondary pulley, by adjusting the pressure supplied to a secondary side cylinder, which is formed of the single hydraulic pressure chamber, independently from performing one of i) control to supply and discharge of the hydraulic fluid and ii) control to adjust the pressure of the hydraulic fluid with respect to the primary side cylinder, wherein a cylinder pressure receiving area of the secondary side cylinder is set such that a safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to a predetermined lowest controllable pressure when the vehicle travels at maximum speed on a level road, is a value less than or equal to 1.5.

Accordingly, the centrifugal hydraulic pressure chamber may be eliminated for all practical purposes, thereby making it possible to provide a vehicular belt-driven continuously variable transmission with a simplified structure.

BRIEF DESCRIPTION OF THE DRAWINGS

The foregoing and further objects, features and advantages of the invention will become apparent from the following description of example embodiments with reference to the accompanying drawings, wherein like numerals are used to represent like elements and wherein:

FIG. **1** is a skeleton view of a vehicular power transmitting device according to one example embodiment of the invention;

FIG. **2** is a clutch and brake application chart showing the operating states of the vehicular power transmitting device shown in FIG. **1**;

FIG. **3** is a sectional view showing part of the structure of an output side variable pulley, which is a constituent member of a belt-driven continuously variable transmission shown in FIG. **1**;

FIG. **4** is a circuit diagram of a hydraulic pressure circuit that forms a hydraulic pressure control apparatus, which supplies hydraulic fluid to the vehicular power transmitting device shown in FIG. **1**;

FIG. **5** is a table listing safety factor calculation results of a vehicle to which the example embodiment has been applied, and the various parameters used in that calculation;

FIG. **6** is a sectional view of a secondary pulley which is a constituent member of a related belt-driven continuously variable transmission; and

FIG. **7** is a circuit diagram of a hydraulic pressure circuit that forms a hydraulic pressure control apparatus according to another example embodiment of the invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. **1** is a skeleton view of a vehicular power transmitting device **10** according to one example embodiment of the invention. The vehicular power transmitting device **10** is an automatic transmission for a transverse mounted engine and may be employed in a FF (front engine, front drive) vehicle. The vehicular power transmitting device **10** includes an engine **12** that serves as a power source for running. Output from the engine **12**, which is an internal combustion engine, is transmitted from a crankshaft of the engine **12** and a torque converter **14**, which is a fluid coupling, to a final reduction gear **22** via a forward-reverse switching apparatus **16**, an input shaft **36**, a belt-driven continuously variable transmitting portion **18**, and a reduction gear device **20**, after which it is distributed to left and right driven wheels **24**L and **24**R. Here the forward-reverse switching apparatus **16** and the belt-driven continuously variable transmitting portion **18** together form a belt-driven continuously variable transmission **30**. Incidentally, the belt-driven continuously variable transmitting portion **18** of this example embodiment may also be regarded as the continuously variable transmitting portion of the invention.

The torque converter **14** is designed to transfer power via fluid and includes a pump impeller **14***p* that is connected to the crankshaft of the engine **12**, and a turbine runner **14***t* that is connected to the forward-reverse switching apparatus **16** via a turbine shaft **34**. Also, a lockup clutch **26** is provided between the pump impeller **14***p* and the turbine runner **14***t*. Switching the supply of hydraulic pressure between an engage side hydraulic fluid chamber and the release side fluid chamber by a switching valve of a hydraulic pressure control apparatus, not shown, or the like engages or releases the lockup clutch **26**. When the lockup clutch **26** is completely engaged, the pump impeller **14***p* and the turbine runner **14***t* rotate together as a single unit. A mechanical oil pump **28** is provided on the pump impeller **14***p*. This mechanical oil pump **28** generates hydraulic pressure used to control the shifting of the belt-driven continuously variable transmitting portion **18**, apply belt squeezing force, and supply lubrication oil to various parts.

The forward-reverse switching apparatus **16** includes a double pinion type planetary gear set as its main component. The turbine shaft **34** of the torque converter **14** is integrally connected to a sun gear **16***s* of the planetary gear set, an input shaft **36** of the belt-driven continuously variable transmitting portion **18** is integrally connected to a carrier **16***c* of the planetary gear set, and the carrier **16***c* and the sun gear **16***s* can be selectively connected together via a forward clutch C**1**. A ring gear **16***r* of the planetary gear set is selectively fixed to a housing via a reverse brake B**1**. The forward clutch C**1** and the reverse brake B**1** are both hydraulic friction engagement devices that are frictionally engaged by a hydraulic cylinder. As shown in FIG. **2**, engaging the forward clutch C**1** and releasing the reverse brake B**1** results in the forward-reverse switching apparatus **16** rotating as a single unit, thus establishing a forward power transmission path such that forward rotation is transmitted to the belt-driven continuously variable transmitting portion **18** without a reduction in speed. On the other hand, by engaging the reverse brake B**1** and releasing the forward clutch C**1**, a reverse power transmission path is established in the forward-reverse switching apparatus **16** so that the input shaft **36** rotates in a direction opposite that of the turbine shaft **34**, which results in reverse rotation being transmitted to the belt-driven continuously variable transmitting portion **18**. Also, releasing both the forward clutch C**1** and the reverse brake B**1** places the forward-reverse switching apparatus **16** in neutral (disconnected state), whereby the transmission of power is interrupted.

The belt-driven continuously variable transmitting portion **18** includes an input side variable pulley **42**, an output side variable pulley **46**, and a transmission belt **48**. The input side variable pulley **42**, provided on the input shaft **36**, is an input side member with a variable effective diameter. The output side variable pulley **46**, provided on the output shaft **44**, is an output side member that also has a variable diameter. The transmission belt **48** serves as a power transmission member that is wound around, in frictional contact with, the variable pulleys **42** and **46** such that power is transmitted via frictional force between the transmission belt and the variable pulleys **42** and **46**. The variable pulley **42** includes a fixed sheave **42***a*, a movable sheave **42***b*, and an input side hydraulic cylinder **42***c*. Similarly, the variable pulley **46** includes a fixed sheave **46***a*, a movable sheave **46***b*, and an output side hydraulic cylinder **46***c*. The fixed sheave **42***a* is fixed to the input shaft **36** while the fixed sheave **46***a* is fixed to the output shaft **44**. The movable sheave **42***b* is provided on the input shaft **36** so as to be able to move in the axial direction but not rotate around its axis relative to the input shaft **36**. Similarly, the movable sheave **46***b* is provided on the output shaft **44** so as to be able to move in the axial direction but not rotate around its axis relative to the output shaft **44**. The input side hydraulic cylinder **42***c* applies thrust that changes the V groove width between the fixed sheave **42***a* and the movable sheave **42***b*, while the output side hydraulic cylinder **46***c* applies thrust that changes the V groove width between the fixed sheave **46***a* and the movable sheave **46***b*. The speed ratio γ (i.e., speed ratio γ=input shaft rotation speed N_{IN}/output shaft rotation speed N_{OUT}) is changed in a continuous fashion by changing the V groove widths of both movable pulleys **42** and **46**, and thus the winding diameter (effective diameter) of the transmission belt **48** around those pulleys, which is done by controlling the hydraulic pressure in the input side hydraulic cylinder **42***c* of the input side variable pulley **42**. Meanwhile, the squeezing force applied to the transmission belt **48** is changed by controlling the hydraulic pressure in the output side hydraulic cylinder **46***c* of the output side variable pulley **46**. The transmission belt **48** is made of left and right steel bands **51**, each of which is formed of a plurality of steel layers, which are fit into multiple metal pieces **49**. Incidentally, the input side variable pulley **42** in this example embodiment may be regarded as the primary pulley of the invention and the output side variable pulley **46** may be regarded as the secondary pulley of the invention. Also, the input side hydraulic cylinder **42***c* of this example embodiment may be regarded as the primary side cylinder of the invention, the output side hydraulic cylinder **46***c* may be regarded as the secondary side cylinder of the invention, and the transmission belt **48** may be regarded as the belt of the invention.

FIG. **3** is a sectional view showing part of the structure of the output side variable pulley **46**, which is a constituent member of the belt-driven continuously variable transmission **30** shown in FIG. **1**. As described above, the output side variable pulley **46** includes the fixed sheave **46***a*, the movable sheave **46***b*, and the input side hydraulic cylinder **46***c*. The fixed sheave **46***a* is integrally formed with the output shaft **44** which is rotatably supported by bearings, not shown, at both ends. The movable sheave **46***b* is fitted onto the output shaft **44** so as to be able to move in the axial direction but not rotate around its axis relative to the output shaft **44**. The output side hydraulic cylinder **46***c* is arranged on the opposite side of the movable sheave **46***b* from the fixed sheave **46***a*. The fixed sheave **46***a* is formed in a circular disc shape protruding in the radial direction and has a fixed side slope face **50** formed on the side opposing the movable sheave **46***b*. The movable sheave **46***b* includes a cylindrical portion **52** fitted onto the output shaft **44** and a disc-shaped flange portion **54** that protrudes in the radial direction from the end portion of the cylindrical portion **52**. A plurality of grooves, not shown, extending in the axial direction are formed in the circumferential direction on both the inner peripheral surface of the cylindrical portion **52** and the outer peripheral surface of the output shaft **44**. These grooves are aligned with each other, i.e., positioned such so as to always be at same phase in the circumferential direction, with ball bearings, not shown, being arranged extending between opposing grooves. Accordingly, the cylindrical portion **52** is able to move smoothly in the axial direction of the output shaft **44** via the ball bearings but is unable to rotate around the output shaft **44**. Also, the flange portion **54** is integrally connected to the cylindrical portion **52** and has a movable side slope face **56** formed on the side opposing the fixed sheave **46***a*. The fixed side slope face **50** and movable side slope face **56** together form a V-shaped groove **58** in which the transmission belt **48** is wound. Here, the angle of inclination, or so-called flank angle, of the fixed side slope face **50** and the movable side slope face **56** is 11 degrees. Also, in FIG. **3**, the portion above the axial center of the output shaft **44** is shown in a state in which the movable sheave **46***b* has been moved to a position closest to the fixed sheave **46***a* side such that the transmission belt **48** is positioned at the outer periphery of the groove **58**, while the portion below the axial center of the output shaft **44** is shown in a state in which the movable sheave **46***b* has been moved to a position farthest from the fixed sheave **46***a* such that the transmission belt **48** is positioned at the inner periphery of the groove **58**.

The output side hydraulic cylinder **46***c* includes a partition **60** fitted onto the output shaft **44** so as to be unable to move in the axial direction of the output shaft **44**, the movable sheave **46***b*, and a spring **62** interposed between the partition **60** and the movable sheave **46***b*. The partition **60** is a cylindrical member, which has a closed bottom at one end, and is fitted onto the output shaft **44** so as to be unable to move in the axial direction relative to the output shaft **44**. This partition **60** includes a first disc portion **60***a* that extends in the radial direction from the outer peripheral surface of the output shaft **44**, a cylinder portion **60***b* that extends in the axial direction toward the movable sheave **46***b* from the outer peripheral end of the first disc portion **60***a*, and a second disc portion **60***c* that extends in the radial direction from one end of the cylinder portion **60***b*. The inner peripheral portion of the first disc portion **60***a* is sandwiched between a step portion formed on the output shaft **44** and a cylindrical spacer **64** fit around the outer peripheral surface of the output shaft **44**, and is thus unable to move in the axial direction. An outer peripheral edge of the second disc portion **60***c* is sealed with an oil tight seal via a seal ring **67** against the inner peripheral surface of a cylindrical outer peripheral cylinder portion **66** provided on the flange portion **54** of the movable sheave **46***b*. Also, the spring **62** is interposed between the first disc portion **60***a* of the partition **60** and the flange portion **54** of the movable sheave **46***b*, which constantly applies thrust to move the movable sheave **46***b* toward the fixed sheave **46***a*. Here, a single hydraulic pressure chamber **68** is formed by the movable sheave **46***b*, the partition **60**, and the output shaft **44**. When a predetermined hydraulic pressure is supplied to this hydraulic pressure chamber **68**, the hydraulic pressure moves the movable sheave **46***b* toward the fixed sheave **46***a* so that it squeezes the transmission belt **48** wound in the groove **58** in the axial direction.

An oil passage **70** that extends in the axial direction is formed inside the output shaft **44** and oil passages **72** and **74** are formed that extend from the oil passage **70** in the radial direction. Also, an oil passage **76** is formed that extends through from the inner periphery to the outer periphery of the cylindrical portion **52** of the movable sheave **46***b*. When hydraulic fluid is supplied to the oil passage **70** while the movable sheave **46***b* is in the state shown in the portion below the axial center in FIG. **3**, that hydraulic fluid passes through the oil passage **72** as well as the oil passage **76** that is connected to the oil passage **72** and into the hydraulic pressure chamber **68**. Thrust from the hydraulic pressure of this hydraulic fluid in addition to the elastic force of the spring **62** move the movable sheave **46***b* toward the fixed sheave **46***a*, thus squeezing the transmission belt in the axial direction. Also, when the movable sheave **46***b* moves into a predetermined position, the hydraulic pressure chamber **68** becomes communicated with the oil passage **74** so that hydraulic fluid is also supplied from this oil passage **74**. The oil passage **70** is connected to a hydraulic pressure circuit **78** of a hydraulic pressure control apparatus **77**, which will be described later.

FIG. **4** is the hydraulic pressure circuit **78** that forms the hydraulic pressure control apparatus **77** for supplying hydraulic pressure to the input side hydraulic cylinder **42***c* and the output side hydraulic cylinder **46***c*.

The hydraulic fluid drawn in from an oil pan **80** via a strainer **82** is pressurized by an oil pump **28** and then supplied to an oil passage **86**. The pressure of the hydraulic fluid in the oil passage **86**, i.e., the pump discharge pressure, is adjusted by a pressure regulating valve **88** that is controlled based on a signal hydraulic pressure output from a solenoid SLT. This adjusted pressure is the line pressure PL. When hydraulic fluid having this line pressure PL is supplied to an oil passage **90** that branches off from the oil passage **86**, it is adjusted by a pressure-regulating valve **92** provided in the oil passage **90**. The pressure-regulating valve **92** is controlled based on a signal hydraulic pressure output from a belt squeeze control solenoid SLS. The pressure adjusted hydraulic fluid passes through the oil passage **70** in the output shaft **44** and is supplied to the output side hydraulic cylinder **46***c*. Accordingly, the belt squeezing force on the transmission belt **48** wound around the input side variable pulley **42** and the output side variable pulley **46** can be adjusted by controlling the pressure of hydraulic fluid supplied to the output side hydraulic cylinder **46***c*.

On the other hand, hydraulic fluid supplied from the oil passage **86** to the oil passage **94** is supplied to a shift speed control valve **96**. The shift speed control valve **96** is switched by a speed increase side solenoid DS**1** and a speed decrease side solenoid DS**2** to open and close communication between a line pressure supply port **98** and a drain port **100**, and an output port **102** to the input side hydraulic cylinder **42***c* of the input side variable pulley **42**. For example, when the speed increase side solenoid DS**1** is on, communication is opened between the line pressure supply port **98** and the output port **102** such that the line pressure PL is supplied to the input side hydraulic cylinder **42***c*. On the other hand, when the speed decrease side solenoid DS**2** is on, communication is opened between the output port **102** and the drain port **100** such that hydraulic fluid is discharged from the input side hydraulic cylinder **42***c*. By controlling the supply and discharge of hydraulic fluid to and from the input side hydraulic cylinder **42***c* in this way, the rotation radius of the transmission belt **48** that is wound around the input side variable pulley **42** is changed appropriately so the belt-driven continuously variable transmitting portion **18** shifts smoothly. Also, as described above, the line pressure in this example embodiment is controlled by the pressure regulating valve **88** via the solenoid SLT, and belt squeezing force control hydraulic pressure P_{OUT} (MPa) supplied to the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* is controlled by the pressure regulating valve **92** via the belt squeeze control solenoid SLS. Both the line pressure and the belt squeezing force control hydraulic pressure P_{OUT} (MPa) can be controlled independently. The various solenoid valves provided in the hydraulic pressure control apparatus **77** are preferably controlled by an electronic control unit based on various specifications provided by a vehicle speed sensor and an accelerator operation amount sensor and the like, not shown.

In this example embodiment, the centrifugal hydraulic pressure canceller chamber **214** shown in FIG. **6** described above is eliminated. Typically when a centrifugal hydraulic pressure canceller chamber is not provided, the belt squeezing force applied to the transmission belt **48** becomes excessive when the vehicle travels at high speeds due to the centrifugal hydraulic pressure generated in the hydraulic pressure chamber **68** shown in FIG. **3**. Here, a safety factor K is used as an index, with respect to belt slip, of the belt squeezing force applied to the transmission belt **48**. This safety factor K is calculated according to Expression (1) below, for example, which is well known.

$$K=\{(P_{OUT}+\beta V^2)S_{OUT}+W\}/(T_{COS}\,\theta/(D\mu)) \qquad (1)$$

Here, P_{OUT} (MPa) represents the belt squeezing force control hydraulic pressure, i.e., the belt tension control hydraulic pressure, supplied to the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c*. β represents the centrifugal hydraulic pressure coefficient (MPa/(km/h)^2) of the output side hydraulic cylinder **46***c*, V (km/h) represents the vehicle speed, S_{OUT} (mm^2) represents the pressure receiving area of the hydraulic pressure chamber **68**, W (N) represents the load of the spring **62**, T (Nm) represents the transfer torque, θ (rad) represents the flank angle of the fixed and movable sheaves **46***a* and **46***b*, D (m) represents the winding diameter of the transmission belt **48** on the input side variable pulley **42** side, and μ represents the coefficient of friction between the transmission belt **48** and the output side variable pulley **46**.

If the safety factor K falls below 1.0, the transmission belt **48** will slip with respect to the output side variable pulley **46**. On the other hand, as the safety factor K increases beyond 1.0, the belt squeezing force applied to the transmission belt **48** becomes excessive, thereby reducing the durability of the transmission belt **48** and reducing belt efficiency. In this case, although there is some variation in the friction coefficient due to the tolerance of the transmission belt **48**, the safety factor K is typically set somewhere within the range of 1.0 to 1.5, inclusive, for example, and preferably in the range of 1.2 to 1.5, inclusive.

Here, in this example embodiment, even if the centrifugal hydraulic pressure canceller chamber of the output side variable pulley **46** is not provided, the belt squeezing force control hydraulic pressure P_{OUT} and the cylinder pressure receiving area S_{OUT} of the hydraulic pressure chamber **68** are set so that the safety factor K falls within the aforementioned range. FIG. **5** is a table listing the calculated results of the safety factor K_0 in a vehicle in this example embodiment, and the various parameters used in that calculation. The safety factor K_0 is calculated while the lowest predetermined pressure that the pressure regulating valve **92** is able to control the belt squeezing force control hydraulic pressure P_{OUT} to (hereinafter also referred to as the "lowest controllable pressure") is supplied to the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* during maximum speed flat road running, i.e., when the vehicle is traveling at maximum speed on a flat road, which is when the affect from centrifugal hydraulic pressure is greatest.

The safety factor K_0 of the vehicle in this example embodiment is set at 1.18, for example. In this case, the safety factor K is a value below 1.5.

Also, the indicated pressure of the belt squeezing force control hydraulic pressure P_{OUT} of the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* during maximum speed flat road running shown in FIG. **5** is 0.327 (MPa), which is above the lowest controllable pressure of 0.2 (MPa). Here, this calculated indicated pressure is calculated assuming that the safety factor K is 1.3 and is thus a hydraulic pressure required to make the safety factor K 1.3. Incidentally, the friction coefficient μ is typically approximately 0.08 to 0.10. In this calculation, the friction coefficient μ is set to be 0.09. Also, the lowest controllable pressure is set based on the specifications of the hydraulic pressure control system provided in each vehicle.

In order to achieve the foregoing safety factor K_0 and indicated pressure, in the vehicle in this example embodiment, the cylinder pressure receiving area S_{OUT} of the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* is set small. In the vehicle in this example embodiment, this pressure receiving area S_{OUT} is set to 121.4 (cm^2). Incidentally, the cylinder pressure receiving area S_{OUT} is set so that the safety factor K does not fall below 1.0 in the context of the maximum transfer torque and the maximum controllable pressure when the vehicle is stopped. Setting the pressure receiving area S_{OUT} small also results in a smaller centrifugal hydraulic pressure coefficient so the safety factor K becomes smaller according to Expression (1). When the pressure receiving area is set small, the line pressure PL required to generate a predetermined belt squeezing force increases, which may adversely affect efficiency due to the increased load on the oil pump **28**. On the other hand, the hydraulic pressure circuit is provided in which the line pressure PL in this example embodiment is regulated by the solenoid SLT and the pressure regulating valve **88**, and the belt squeezing force control hydraulic pressure P_{OUT} supplied to the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* is regulated by the belt squeeze control solenoid SLS and the pressure regulating valve **92**. Because the line pressure PL and the belt squeezing force control hydraulic pressure P_{OUT} may be regulated separately, an increase in the line pressure PL can be kept to a minimum. That is, an increase in the line pressure PL is limited to the speed reduction range (speed ratio $\gamma>1.0$) where the hydraulic pressure supplied to the output side hydraulic cylinder **46***c* becomes higher than the hydraulic pressure supplied to the input side hydraulic cylinder **42***c*, and thus limited to times such as during take-off from a standstill or a kickdown while traveling at low speeds. Therefore, by keeping an increase in the line pressure PL to a minimum, an adverse affect on the practical fuel consumption is inhibited.

Also in the vehicle in this example embodiment, the lowest controllable pressure is set low. More specifically, in the vehicle in this example embodiment, the lowest controllable pressure is 0.2 (MPa), as shown in FIG. **5**. Accordingly, the indicated pressure of the vehicle in this example embodiment of 0.327 (MPa) exceeds 0.2 (MPa), meaning that it can be controlled. When air enters the hydraulic pressure chamber **68** when the hydraulic pressure in the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* is drained, the responsiveness when hydraulic pressure is supplied decreases. So, it is necessary to apply the hydraulic pressure required to fill the hydraulic pressure chamber **68** with hydraulic fluid. The required hydraulic pressure is the lowest controllable pressure, but in this example embodiment a hydraulic pressure control valve that controls the pressure to an extremely low pressure is provided to lower the lowest pressure. Also, control to reduce the variation in the control valve, such as the hydraulic pressure learning using the hydraulic pressure sensor or the like, is performed.

Also in the vehicle in this example embodiment, in order to keep the centrifugal hydraulic pressure to a minimum, the rotational speed of the output side variable pulley **46** is set relatively low. In a structure such as that of the vehicle in this example embodiment in which rotation from the engine **12** is input to the belt-driven continuously variable transmitting portion **18** without a reduction in speed, and in which the reduction gear device **20** is arranged after the belt-driven continuously variable transmitting portion **18**, the rotation speed of the output side variable pulley **46** with respect to the vehicle speed is determined by the reduction gear ratio of that reduction gear, device **20** and the tire radius. Accordingly, the rotation speed may be reduced by decreasing the reduction gear ratio or increasing the tire radius.

Further, in the vehicle in this example embodiment, an engine **12** is used that can generate relatively large transfer torque T. As a result, the safety factor K can be set low. Taking all of these factors into account, the safety factor K can be kept down to the safety factor K_0 described above, and a pressure equal to or greater than the lowest controllable pressure can always be maintained.

Accordingly, although the belt squeezing force would become excessive at maximum speed due to increased thrust generated by centrifugal hydraulic pressure pushing the movable sheave **46***b* of the output side movable pulley **46** in the direction that increases the belt squeezing force because no centrifugal hydraulic pressure canceller chamber is provided to cancel out that thrust, according to the belt-driven continuously variable transmission **30** of this example embodiment, the belt squeezing force is inhibited from becoming excessive by reducing the pressure receiving area S_{OUT} of the output side hydraulic cylinder **46***c*. In this case, it is necessary to also reduce the pressure supplied to the output side hydraulic cylinder **46***c*. Regarding this point, controlling the line pressure PL and the belt squeezing force control hydraulic pressure P_{OUT} in the output side hydraulic cylinder independently makes it possible to avoid problems such as the line pressure PL, together with the belt squeezing force control hydraulic pressure P_{OUT} in the output side hydraulic cylinder **46***c*, becoming too low, or not being able to shift to increase the speed due to the hydraulic pressure necessary to shift the belt-driven continuously variable transmitting portion **18** not being supplied to the input side hydraulic cylinder **42***c* or the like. Also, when the pressure receiving area S_{OUT} of the output side hydraulic cylinder **46***c* is reduced, the line pressure PL must be increased to increase the hydraulic pressure supplied to the output side hydraulic cylinder **46***c* when the vehicle is traveling at low speeds. However, if the line pressure PL can be controlled independently from the belt squeezing force control hydraulic pressure P_{OUT} in the output side hydraulic cylinder **46***c*, an increase in the line pressure PL can be limited to roughly the speed reduction range ($\gamma>1$) so adverse affects on practical fuel consumption can be avoided. Also, the cylinder pressure receiving area S_{OUT} of the output side hydraulic cylinder **46***c* is reduced until the safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the output side hydraulic cylinder **46***c* is set to the lowest pressure during maximum speed flat road running, in which the vehicle is running at maximum speed on a flat road, becomes a value less than or equal to 1.5, which makes it possible to keep the durability of the belt from declining. As a result, the centrifugal hydraulic pressure canceller chamber can be eliminated for all practical purposes, while avoiding the problems of the related art described above.

Also, the belt-driven continuously variable transmission **30** of this example embodiment is lighter, more compact, and less expensive because the centrifugal hydraulic pressure canceller chamber is eliminated. Also, there is no need for the hydraulic fluid that was supplied to the centrifugal hydraulic pressure canceller chamber so the volume of the oil pump **28** can be reduced.

Continuing on, another example embodiment of the invention will now be described. Parts in this example embodiment that are the same as parts in the example embodiment described above will be denoted by the same reference characters and descriptions thereof will be omitted.

FIG. **7** is a circuit diagram of a hydraulic pressure circuit **302** that forms a hydraulic pressure control apparatus **300** according to another example embodiment of the invention. The structure aside from the hydraulic pressure control apparatus **300** is the same as it is in the vehicular power transmitting device **10** described above so a description thereof will be omitted.

The hydraulic fluid drawn in from the oil pan **80** via the strainer **82** is pressurized by the oil pump **28** and then supplied to an oil passage **304**. The pressure of the hydraulic fluid in the oil passage **304**, i.e., the pump discharge pressure, is adjusted by a pressure-regulating valve **306**. The adjusted pressure is the line pressure PL. The hydraulic fluid having this line pressure PL is supplied to oil passages **310** and **312** that branch off from a branching point in the oil passage **308**. The pressure of hydraulic fluid supplied to the oil passage **310** is adjusted by a pressure-regulating valve **314**. The pressure-regulating valve **314** is controlled based on a signal hydraulic pressure output from an input side hydraulic pressure control solenoid SLP. The pressure adjusted hydraulic fluid is then supplied to the input side hydraulic cylinder **42***c* of the input side variable pulley **42**.

On the other hand, the pressure of the hydraulic fluid supplied to the oil passage **312** is adjusted by a pressure-regulating valve **316**. The pressure regulating valve **316** is controlled based on a signal hydraulic pressure output from an output side hydraulic pressure control solenoid SLS. The pressure adjusted hydraulic fluid is then supplied to the output side hydraulic cylinder **46***c* of the output side variable pulley **46**.

Also, the signal hydraulic pressures output from the input side hydraulic pressure control solenoid SLP and the output side hydraulic pressure control solenoid SLS are input to a three-way switching valve **318**. This three-way switching valve **318** is switched by the input side hydraulic pressure control solenoid SLP and the output side hydraulic pressure control solenoid SLS to open and close communication between a first input port **320** and a second input port **322**, and an output port **324**. For example, when the input side hydraulic pressure control solenoid SLP is on, communication is opened between the first input port **320** and the output port **324** such that the signal hydraulic pressure of the input side hydraulic pressure control solenoid SLP is input as the pilot pressure of the pressure regulating valve **306**. On the other hand, when the output side hydraulic pressure control solenoid SLS is on, communication is opened between the second input port **322** and the output port **324** such that the signal hydraulic pressure of the output side hydraulic pressure control solenoid SLS is input as the pilot pressure of the pressure regulating valve **306**. Accordingly, the line pressure PL is controlled according to the magnitude relation between the signal hydraulic pressures of the input side hydraulic pressure control solenoid SLP and the output side hydraulic pressure control solenoid SLS, and a higher hydraulic pressure is supplied to the pressure regulating valve **306**. Also, the pressure-regulating valve **306** is controlled by the higher hydraulic pressure to regulate the line pressure PL. On the other hand, the belt squeezing force control hydraulic pressure P_{OUT} supplied to the hydraulic pressure chamber **68** of the output side hydraulic cylinder **46***c* is regulated by the pressure regulating valve **316** via the output side hydraulic pressure control solenoid SLS, thus the line pressure PL and the belt squeezing force control hydraulic pressure P_{OUT} are able to be controlled independently.

This kind of hydraulic pressure circuit **302** is also able to achieve the same effects as those achieved in the example embodiment described above, and thus enables the centrifugal hydraulic pressure canceller chamber to be eliminated for all practical purposes.

Heretofore, example embodiments of the invention have been described in detail with reference to the accompanying drawings. Other example embodiments of the invention are also possible.

For example, in the hydraulic pressure circuits **78** and **302** in the foregoing example embodiments, the line pressure PL and the belt squeezing force control hydraulic pressure P_{OUT}, supplied to the hydraulic pressure chamber **68** of the output side variable pulley **46**, may be controlled independently. However, as long as the hydraulic pressures are independently controllable, the invention may also be applied to a hydraulic pressure circuit having another structure.

Also, in these example embodiments, the belt-driven continuously variable transmitting portion **18** is shifted by controlling the supply and discharge of hydraulic pressure to and from the input side hydraulic cylinder **42***c*. However, the invention may also be applied to a structure in which a belt-driven continuously variable transmitting portion is shifted by controlling the pressure of the hydraulic fluid supplied to the input side hydraulic cylinder **42***c*.

Also, the vehicular power transmitting device **10** in these example embodiments is applied to an FF (front engine, front drive) type vehicle, but the invention may also be applied to another type of vehicle such as a four-wheel-drive vehicle. Further, the structure and the like of the forward-reverse switching apparatus **16** may be freely modified in a manner consistent with the scope of invention.

While the invention has been described with reference to example embodiments thereof, it is to be understood that the invention is not limited to the described embodiments or constructions. To the contrary, the invention is intended to cover various modifications and equivalent arrangements. In addition, while the various elements of the described embodiments are shown in various example combinations and configurations, other combinations and configurations, including more, less or only a single element, are also within the spirit and scope of the invention.

The invention claimed is:

1. A method of controlling a vehicular belt-driven continuously variable transmission for use in a vehicle traveling at speeds up to a maximum speed, the belt-driven continuously variable transmission including a continuously variable transmitting portion into which rotation from a power source is input without a reduction in speed while a vehicle is traveling forward, and a single hydraulic pressure chamber provided for a secondary pulley, the method comprising:

shifting the continuously variable transmitting portion by controlling one of i) the supply and discharge of a hydraulic fluid and ii) the pressure of the hydraulic fluid with respect to a primary side cylinder provided for a primary pulley; and

adjusting belt squeezing force of a belt, which is wound around the primary pulley and the secondary pulley, by adjusting the pressure supplied to a secondary side cylinder, which is formed of the single hydraulic pressure chamber, independently from performing one of i) control to supply and discharge of the hydraulic fluid and ii) control to adjust the pressure of the hydraulic fluid with respect to the primary side cylinder,

wherein performing the control to adjust the pressure of the hydraulic fluid includes adjusting a first pressure regulating valve and wherein adjusting the pressure supplied to the secondary side cylinder includes adjusting a second pressure regulating valve; and

wherein a cylinder pressure receiving area of the secondary side cylinder is set such that a safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to a predetermined lowest controllable pressure when the vehicle travels at maximum speed on a flat road, is a value less than or equal to 1.5.

2. The method of controlling a vehicular belt-driven continuously variable transmission according to claim **1**, wherein performing the control to supply and discharge of the hydraulic fluid includes adjusting a shift speed control valve.

3. The method of controlling a vehicular belt-driven continuously variable transmission according to claim **1**, wherein when the speed of the vehicle is increased, the line pressure is supplied to the primary side cylinder.

4. A vehicular belt-driven continuously variable transmission for use in a vehicle traveling at speeds up to a maximum speed, the belt-driven continuously variable transmission comprising:

a continuously variable transmitting portion into which rotation from a power source is input without a reduction in speed while a vehicle is traveling forward;

a single hydraulic pressure chamber provided for a secondary pulley; and

a hydraulic pressure control apparatus that is configured to shift the continuously variable transmitting portion and configured to adjust a belt squeezing force of a belt, which is wound around a primary pulley and the secondary pulley;

wherein the hydraulic pressure control apparatus shifts the continuously variable transmitting portion by controlling one of i) the supply and discharge of a hydraulic fluid and ii) the pressure of the hydraulic fluid with respect to a primary side cylinder provided for the primary pulley,

wherein the hydraulic pressure control apparatus adjusts the belt squeezing force of the belt by controlling the pressure supplied to a secondary side cylinder formed of the single hydraulic pressure chamber,

wherein the hydraulic pressure control apparatus controls a line pressure and the pressure supplied to the secondary side cylinder independently, the hydraulic pressure control apparatus including a first pressure regulating valve configured to control the line pressure and a second pressure regulating valve configured to control the pressure supplied to the secondary side cylinder, and

wherein a cylinder pressure receiving area of the secondary side cylinder is set such that a safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to a predetermined lowest controllable pressure when the vehicle travels at maximum speed on a flat road, is a value less than or equal to 1.5.

5. The vehicular belt-driven continuously variable transmission according to claim **4**, wherein the cylinder pressure receiving area of the secondary side cylinder is set such that the safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to the predetermined lowest controllable pressure when the vehicle travels at maximum speed on a flat road, is a value between 1.0 and 1.5, inclusive.

6. The vehicular belt-driven continuously variable transmission according to claim **5**, wherein the cylinder pressure receiving area of the secondary side cylinder is set such that the safety factor, with respect to belt slip, of the belt squeezing force obtained when the pressure supplied to the second side cylinder is set to the predetermined lowest controllable pressure when the vehicle travels at maximum speed on a flat road, is a value between 1.2 and 1.5, inclusive.

7. The vehicular belt-driven continuously variable transmission according to claim **4**, wherein the hydraulic pressure control apparatus further includes a shift speed control valve.

8. The vehicular belt-driven continuously variable transmission according to claim **4**, wherein when the speed of the vehicle is increased, the line pressure is supplied to the primary side cylinder.

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